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Procedia CIRP 41 (2016) 812 – 817

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48th CIRP Conference on MANUFACTURING SYSTEMS - CIRP CMS 2015

Approach for a general pose-dependent model of the dynamic behavior of large lightweight machine tools for vibration reduction

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Abstract

Large-scale machine tools for the manufacturing of large work pieces, e.g. blades, casings or gears for wind turbines, feature pose-dependent dynamic behavior. Small structural damping coefficients lead to long decay times for structural vibrations that have negative impacts on the production process. Typically, these vibrations are handled by increasing the stiffness of the structure by adding mass. That is counterproductive to the needs of sustainable manufacturing as it leads to higher resource consumption both in material and in energy. Recent research activities have led to higher resource efficiency by radical mass reduction that rely on control-integrated active vibration avoidance and damping methods. These control methods depend on information describing the dynamic behavior of the controlled machine tools in order to tune the avoidance or reduction method parameters according to the current state of the machine.

The paper describes the approach for a general pose-dependent model of the dynamic behavior of large lightweight machine tools that provides the necessary input to the aforementioned vibration avoidance and reduction methods to properly tackle machine vibrations. The paper starts with an overview of the state of the art of the pose-dependent dynamic behavior of machine tools followed by the most common methods for vibration avoidance and reduction. Based on the results of an experimental modal analysis of a lightweight machine tool structure, the relevant pose-dependency is shown and the relevant parameters to derive the dynamic behavior are deduced. Then, a general model structure to model the machine tool's dynamic behavior is introduced. After updating the model parameters to different discrete machine poses the dynamical behavior of the model and the real machine tool structure are compared. Finally, it is explained how the model contributes to the actual vibration reduction of lightweight machine tools.

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Keywords: Machine Tool Dynamics; Modelling; Pose-Dependency

1. Introduction

New paradigms in manufacturing promote the design of resource-efficient machines. Non-negligible lightweight construction causes lower passive stiffness of the structural machine components due to mass reduction. However, low passive stiffness causes lower structural eigenfrequencies [1] and higher decay times for vibrations. Producing accurate parts on these machine structures, active control strategies are necessary to compensate the missing passive stiffness – either, with advanced control strategies or by applying active vibration reduction methods. However, the structural dynamics of each machine tool are varying within the machine's working space [2] and therefore, are pose-dependent. In large lightweight machine tools this effect even increases. Controlling a feed drive effectively with flanged lightweight structure, the dominant eigenfrequencies within the drive's bandwidth have to be known for tuning diverse active vibration reduction methods (e.g. continuously updating filter parameters). Therefore, the actual machines' eigenfrequencies for each position within the working space has to be known exactly. Or, an accurate machine model has to be available, which

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doi:10.1016/j.procir.2015.12.014

represents the actual dynamic behavior of a machine structure. Otherwise, vibrations cannot be fully eliminated or suppressed by active control strategies.

Thus, the goal of this paper is to introduce an approach for a general pose-dependent model of the dynamic behavior of large lightweight machine tools for vibration reduction. There, a physical machine model is adapted on the actual dynamic behavior of a machine tool by online parameter updating algorithms using machine and additional sensor signals. Section 2 gives a short overview on vibrations, popular vibration reduction methods and on current approaches to model the dynamic behavior of machine tools. This is followed by the approach overview with requirements and constraints in section 3. After introducing a laboratory prototype in section 4, the general machine model based on that prototype is derived in section 5. In section 6, the general machine model is verified. The discussion and outlook in section 7 finalize the paper.

2. State of the Art

Vibrations which generally occur on machine tools are externally excited and self-generative vibrations. Externally excited vibrations are primarily caused by positioning movements, vibrations of the base or machining processes. Acceleration and deceleration of the axes as well as shocks excite the machine structure transiently in its weakly damped natural frequencies. These are free vibrations, which decay depending on the damping ratio of the machine structure. Machining processes excite forced vibrations with the tooth passing frequency. Chattering as one possibility of self-generative vibrations is produced by interaction of the machining process with the flexible machine structure [3]. Because of the complexity of self-excited vibrations, in a first step just forced vibrations are considered for the stated approach within the paper. The vibrations due to the machining process are mostly vibrations of the spindle and tool holder as well as the work piece mount in the area of several hundred to several thousand Hertz. Structural vibrations usually range to about 120 Hz [4].

To handle vibrations in machine tools effectively, vibration reduction is inevitable. Fig. 1 classifies the most common vibration reduction methods. Here, vibration reduction is divided into excitation avoidance and vibration damping. Excitation avoidance considers the avoidable vibration sources based on the interaction of acceleration and

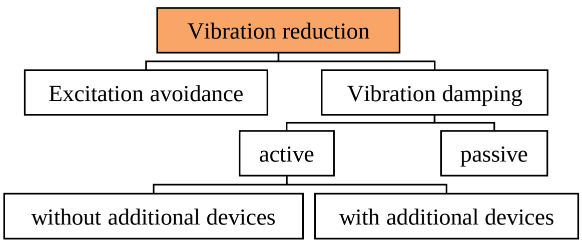

Fig. 1. Possibilities of vibration reduction in machine tools

deceleration forces with the machine structure during axes motion. Possibilities for excitation avoidance are using motion profiles based on continuous [5] or adjusted jerk profiles [6], set-point filtering and input shaping [5]. Besides the motion profile also the path profile imprints vibrations into the machine tool structure if the path curvature is not continuous. In order to avoid this various spline interpolations as, for example, the Cornu spline, are developed [7].

Vibration damping is divided into active and passive methods. The active methods regarded here can further be subdivided into methods with or without additional devices. In the latter case this is mostly a drive-based vibration reduction. [5], for example, measures the disturbing vibration and generates synthetic counteracting velocity signal via the feed drive. Active vibration damping with additional devices is often done via inertial mass dampers [8] or actuators with advanced control algorithms [9]. For an efficient use of especially the excitation avoidance and drive based vibration reduction methods (e.g. for tuning filter constants and generating counteracting signals) the knowledge on the vibrations' eigenfrequencies for every set of condition of the machine tool is necessary.

Regarding typical industrial machine tools the eigenfrequencies of the machine tool's structure and each feed drive control loop are often known just for a special set of conditions. A complete measurement in all positions and under different conditions is extremely cost-intensive, a detailed pose-dependent dynamic machine model is not state of the art. Research focusses on the real-time computation and simulation of pose-dependent dynamic machine tool behavior. [10] models a flexible machine tool on several positions beforehand and uses those linear models to predict the pose-dependent system behavior. A position-dependent, substructurally synthesized machine model of reduced order for structural design modifications and topology optimization is created by [2] to achieving targeted productivity by considering the position-dependent process-machine interactions. [11] also follows a substructurally reduced order approach based on detailed finite element models. [10] and [2] evaluate their models in discrete positions, [2] even with process forces. However, the number of evaluated discrete positions is low and therefore, the gained accuracy of the prediction over the entire machine working space limited. The model by [11] considers a nearly continuous movement over the working space, but no machining force interaction or disturbances.

The physical modelling of machine tools presumes detailed knowledge of parameters, which are normally not exactly known. Furthermore, it is not guaranteed that the model reflects the real dynamic machine tool behavior as long as the model is not validated by the measurement of the real machine tool utilised. These measurements would need to be done in different poses and under different machining scenarios. Even so, the medium- and long-term change of the dynamic behavior or variances between identically constructed machines [12] are still not considered.

3. Approach overview with requirements and constraints

The goal of this paper is to present an alternative approach, where physical and experimental parametric modelling of machine tools are combined (Fig. 2). There, general parametric machine models for large lightweight machine tool structures are derived. The options for machine kinematics for large work pieces are limited. Because of the space available and the mass ratios, kinematics with just tool-sided movement are favorable [1]. So, for the generalized parametric machine model, only gantry-type or travelling column machine tool kinematics are considered. The model parameters are updated online depending on the actual machine tool dynamic behavior (see Fig. 2). The parameter adjustments due to changes in the pose of the machine tool structure are of central importance. Therefore, internal measurement systems (positions, velocity, forces and moments) as well as additional external sensors (e.g. acceleration) are used. The general machine model, which depicts the actual dynamic behavior of the large lightweight machine tool structure can be applied e.g. in vibration reduction methods.

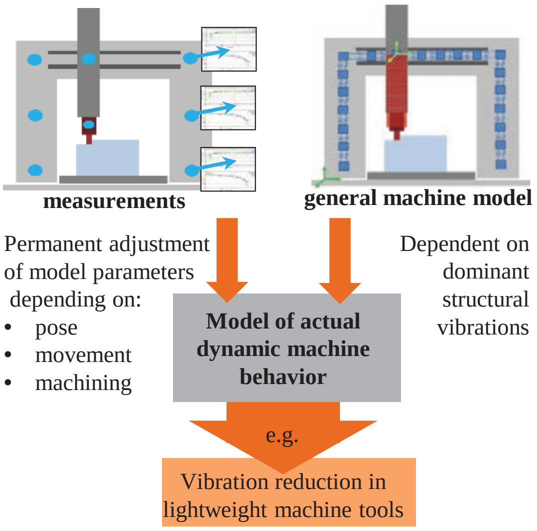

Fig. 2. Approach of the general machine model

Because of realtime-capability and simple applicability, the general machine model is to be modelled as simply as possible. But, the adjusted model has to fit the dynamic machine tool behavior within defined boundaries. The necessary accuracy of the adjusted model is defined by the vibration reduction method the model knowledge is used for. Besides that, the vibration reduction method defines the following points:

- Kind and amount of eigenmodes which have to be depicted by the general machine model. These modes are controlled afterwards.
- Frequency range of interest.
- Modelling of the compliance by a finite segment (FS) approach is sufficient or a flexible multi-body system (FMBS) approach is necessary.

As the general machine model is very simple, but the dynamic behavior of the real machine changes permanently within the working space and under real machining conditions, the parameters of the general machine model need to be adjusted online. The used updating method defines the specific structure of the general machine model also:

- Amount of parameters to be updated: Depending on the amount of bodies and the modelling strategy of compliances (FS or FMBS approach). This defines the amount of necessary input signals for parameter updating.
- Consideration of axes' movement within the model: This defines if the model is invariant or variant in its structure.
- Storage of the model knowledge and usage of previously aquired knowledge.
- Real-time capability.

The structure of the equation of movement of the model and the dependency of the parameters to be updated is most relevant for the definition of the parameter updating method. This defines the system of equations for the calculation of the unknown parameters.

Generally, the design with cantilever structures of gantry-type or travelling column machine tools predefines the dominant eigenfrequencies and eigenmodes. Ideally, one general machine model for each type of machine would be sufficient. But, at first instance a laboratory prototype of a lightweight machine tool structure is exemplarily used to model the general machine model for a travelling column machine. However, measurements on a commercial, large scale machine tool are in the planning.

4. Examination of a laboratory prototype of a lightweight machine structure

The following laboratory prototype of a lightweight machine tool structure (Fig. 3) is examined for investigating the stated approach. The lightweight machine tool structure represents a travelling column machine kinematics where the ram is moving within two support structures which model the travelling column.

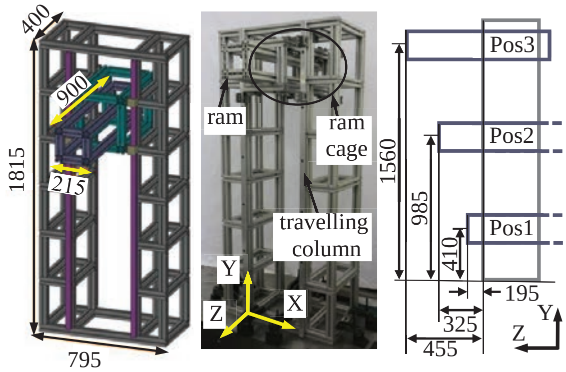

Fig. 3. Laboratory prototype of a lightweight machine tool. (left: CAD-model (dimensions in mm); middle: real prototype; right: investigated poses)

The laboratory prototype is made of modular aluminum strut profiles and the corresponding connectors. Dimensions and masses are depicted in Fig. 3 and Table 1. The laboratory prototype is a passive machine structure and contains no drives. The ram is vertically and horizontally adjustable in each position by clamping. The stiffness increasing components like a spindle and guide rails are replicated. Like in a real machine tool, the ram has a vertical degree of freedom (dof) respective to the travelling column. This dof is taken by the clamping of the ram at the spindle replicate.

Table 1. Technical data of the laboratory prototype

	travelling column	ram cage	ram
material	Aluminum	Aluminum	Aluminum
mass [kg]	80	12.3	8.6

The pose-dependent dynamic behavior was examined by experimental modal analysis at nine discrete poses in the working space of the structure. Eigenmodes up to 250Hz were measured. However, the dominant eigenmodes are the first bending mode in X-direction, the first bending mode in Z-direction and the first torsion mode around the Y-axis. The results for the exemplary poses shown in Fig. 3 are summarized in Table 2. These poses cover the maximum frequency range of the particular eigenmodes of the laboratory prototype.

Table 2. Results of the modal analysis of the laboratory prototype.

Ram position	1 st bending mode X [Hz]	1 st bending mode Z [Hz]	1 st torsion mode Y [Hz]
Pos 1	15.93	22.75	23.74
Pos 2	13.72	19.09	20.50
Pos 3	11.87	15.70	17.55
Deviation	25%	31%	26%

From Table 2 it can be conclude that the eigenfrequencies of the laboratory prototype are pose-dependent. The first three dominant eigenfrequencies have a deviation between 25% and 30% relating to the highest eigenfrequency of the particular eigenmode. This dominant dynamic behavior has to be represented within the general machine model.

5. General Machine Model of the Laboratory Prototype

Based on this previous knowledge and in order to use the model for drive-based vibration reduction a general parametric machine model for the laboratory prototype is set up. The parameterized model is implemented in Matlab using the symbolic math toolbox. Describing the relevant eigenmodes of the laboratory prototype, but keeping the elastic degrees of freedom (dof) at a minimum, an approach with two rigid bodies (amount of bodys $p = 2$) for the general machine model is used. The mechanical analogous model is shown in Fig. 4.

Body1 with the mass m_1 stands for the travelling column, the body2 with mass m_2 for the moveable ram. Body1 has three elastic rotational dofs respectively to a fixed fundament. These dofs simulate the dominant eigenfrequencies

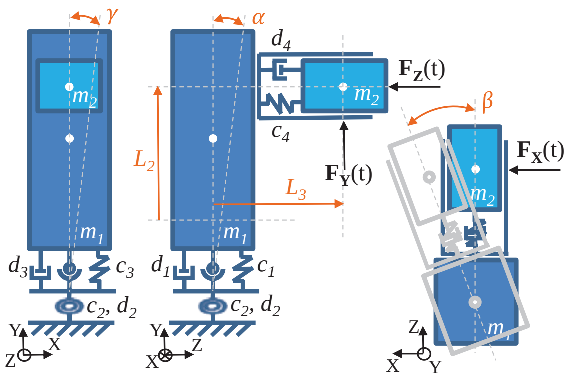

Fig. 4. Mechanic analogous model of the general machine model

of the travelling column from the experimental modal analysis. Therefore, the elasticity of the real travelling column is modelled with a simple finite segment approach using one rigid mass and three springs for each dof. The spring c_1 influences the bending mode in Z-direction, c_2 the torsion-mode around the Y-axis and c_3 the bending-mode in X-direction. The dampers d_i for $i = 1,2,3$ determine the resonance magnification, but have no influence on the eigenfrequencies. For now, we neglect a translational movement of the travelling column, as this movement has no influence on the pose-dependency of the dynamic behavior of the laboratory prototype. Body2 has a rigid translational dof in vertical Y-direction to simulate the movement of the ram and an elastic dof determined by c_4 und d_4 in horizontal Z-direction. At the ram, the external forces F_X, F_Y and F_Z are applied, which e.g. result from the machining process.

The derivation of the equation of motion for the mechanical analogous model in Fig. 4 the Newton-Euler equations for the generalized coordinates $\boldsymbol{y} = [\alpha, \beta, \gamma, L_2, L_3]^T$ (dimension $f x 1$ with $f = 5$) are formulated. For eliminating the reaction forces and solving the Newton-Euler equations, the method of D'Alembert in the Lagrangian description is applied [13]. This results in the following equation of motion for the holonomic multi-body sytem (MBS) from Fig. 4. Equation (2) is the generalized representation of equation (1). The used symbols are explained in Table 3.

$$\sum_{i=1}^{p}\left[\mathbf{J}_{Ti}^{T} m_i \mathbf{J}_{Ti} + \mathbf{J}_{Ri}^{T}\mathbf{I}_i\mathbf{J}_{Ri}\right]\ddot{\mathbf{y}} + \sum_{i=1}^{p}\left[\mathbf{J}_{Ti}^{T} m_i \bar{\mathbf{a}}_i + \mathbf{J}_{Ri}^{T}\mathbf{I}_i\bar{\boldsymbol{\alpha}}_i + \mathbf{J}_{Ri}^{T}\tilde{\boldsymbol{\omega}}_i\mathbf{I}_i\boldsymbol{\omega}_i\right] = \sum_{i=1}^{p}\left[\mathbf{J}_{Ti}^{T}\mathbf{f}_i^{e} + \mathbf{J}_{Ri}^{T}\mathbf{l}_i^{e}\right] \tag{1}$$

$$\mathbf{M}(y,t)\ddot{\mathbf{y}} + \mathbf{k}(y,\dot{y},t) = \mathbf{q}(y,\dot{y},t) \tag{2}$$

with $\mathbf{M} \in \mathbb{R}^{f x f}$ and $\mathbf{k}, \mathbf{q} \in \mathbb{R}^{f}$

For the MBS (Fig. 4) with the generalized coordinates $\boldsymbol{y}$ a system of f second order ordinary differential equations (ODE) is obtained. The movement of the ram results in a variable Inertia $\boldsymbol{I}$ and variable moments $\boldsymbol{l}_i^e$ on body1. As we consider rigid bodies with lumped masses in the center of mass, the

Table 3. Symbol explanation for the equation of motion for the general machine model

Symbol	Explanation
$\mathbf{J}_{Ti}, \mathbf{J}_{Ri}$	Jacobian matrix of the translation and rotation of body i
$\mathbf{I}_i$	Inertia of body i
$\bar{\boldsymbol{a}}_i$	Local acceleration
$\bar{\boldsymbol{\alpha}}_i$	Local angular acceleration
$\boldsymbol{\omega}_i$	Angular velocity
$\tilde{\boldsymbol{\omega}}_i$	Skew symmetric matrix of rotation ($\tilde{\omega}\, a = \omega \times a$)
$\mathbf{f}_i^e, \mathbf{l}_i^e$	Active forces and moments
$\mathbf{M}$	Generalizes mass matrix
$\mathbf{k}$	Vector of the generalized centrifugal, gyroscopic and Coriolis forces
$\mathbf{q}$	Vector of the generalized active forces.

active forces are not pose-dependent. The other parameters, like the Jacobian matrixes and the local velocities and accelerations, contain position vectors of the bodies and must be, therefore, pose-dependent as well.

The pose-dependency of the gained MBS model is shown in Fig. 5. There the movement of body1 is depicted while the ram is moving in positive Z-direction. The system was excited by a step of F_Z. We see that the higher the ram moves, the lower the frequency of the vibrations of body1.

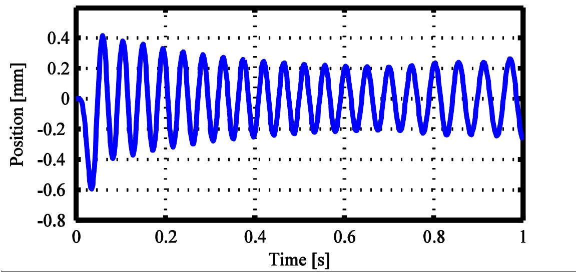

Fig. 5. Pose-dependent dynamic behavior of the MBS-model. (Left: Position of Body1; Right: Movement of the ram)

To describe the actual dynamic behavior of the laboratory prototype, the model parameters have to be defined, calculated and identified for the actual state. The masses of the bodies m_1 and m_2 can be determined beforehand and are constant for the model on hand. The inertia $\boldsymbol{I}$ can be calculated with the knowledge of the axes measuring system. The Jacobian matrixes $\mathbf{J}_{Ti}, \mathbf{J}_{Ri}$ as well as the local velocities and accelerations yield from system theoretical considerations. The model parameters c_i and d_i for $i = 1, \ldots, 4$ are unknown and need to be identified for the actual machine state. External forces must be measured. Before designing and implementing the parameter updating algorithms the general machine model has to be verified for the laboratory prototype. It has to be tested, if it is possible to adapt the dynamic model behavior to the measurement just by tuning the model parameters.

6. Verification of the general machine model for the laboratory prototype

Qualitatively, the dominant dynamic behavior of the laboratory prototype is representable with the derived general machine model (Fig. 5). Within this section, the quantitative agreement between model and real prototype is investigated when the model parameters are adjusted to the real behavior.

At this point, the parameter adjustment is done by hand. The masses and inertia factors are calculated with the technical data of the laboratory prototype. The model parameters c_i and d_i for $i = 1, \ldots, 4$ are adjusted for each pose, so that the acceleration frequency response function (FRF) of the model and the FRFs gained in the experimental modal analysis match. The model FRF is built by the DFTs (discrete Fourier transform) of the force impulse input and the acceleration output of a representing point on body1. Fig. 6 and Fig. 7 show the sum FRFs of responses in X- and Z-direction in different positions. In Fig. 6, the blue sum FRF illustrates the eigenfrequencies of the laboratory prototype up to 250Hz in Pos3. The first three eigenfrequencies are the first bending mode in X-direction, the first bending mode in Z-direction and the first torsion mode around the Y-axis (compare Table 2). These are the dominant eigenmodes of the laboratory prototype. Higher eigenmodes appear in frequencies over 60Hz. The red model sum FRF matches the blue measured one within minimal deviation. All three eigenfrequencies are clearly visible as resonances. Fig 7 shows the zoomed details of the sum FRFs in Pos2 and Pos1.

In both subfigures the measurement (blue) shows that the eigenfrequencies of the first bending mode in Z-direction and the first torsion mode in Y-direction are not clearly separated in terms of resolution. The lower the position of the ram, the more these eigenfrequencies merge. The model FRF (red) fits the visible resonances in both cases very well. The middle not quite clear resonances are depicted by the model as well. These resonances are seen in the model FRF as antiresonances.

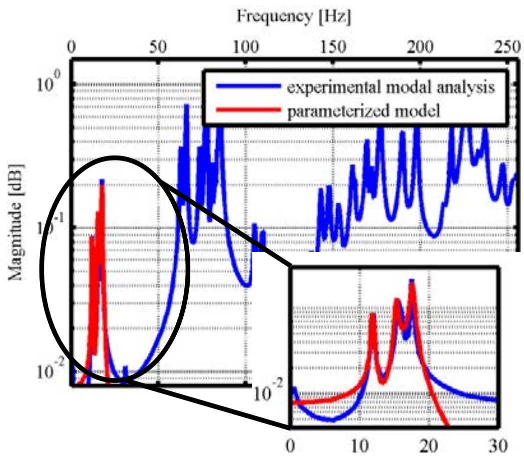

Fig. 6. Comparison of the sum FRF between model and measurement in Pos3

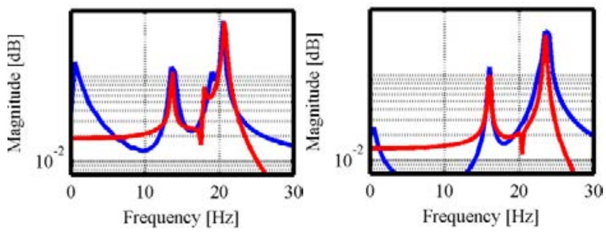

Fig. 7. Comparison of the sum FRF between model and measurement in Pos2 (left) and in Pos1 (right).

The examination of the model and the comparison of the FRFs of model and measurement show, that the derived general machine model of a travelling column machine can represent the dynamic behavior of the laboratory prototype within minimum deviation by just adapting the model parameters. The goal to depict the first three dominant eigenfrequencies and – modes in different poses by simply identifying the model parameters c_i and d_i for $i = 1, \ldots, 3$ is reached.

7. Discussion and outlook

The paper presents a new approach, where physical and experimental parametric modelling of machine tools are combined. After an introduction into the topic, the state of the art of vibration reduction and the modelling of the pose-dependent dynamic machine tool behavior is presented. Following, the idea of a general machine model with parameters adapted online depending on the actual pose and dynamic behavior of the machine tool is explained. On basis of the investigation of a laboratory lightweight machine tool structure a general machine model for a travelling column machine is derived. Finally, it is verified that the parameters of the derived model can be adapted, so that the sum FRFs of the model fits to the sum FRFs from measurement in different poses.

It is proven, that the stated approach is applicable. The pose-dependent dynamic behavior of large lightweight machine tools can be represented by a general machine model which parameters are updated depending on the actual pose. For now, the model parameters are tuned by hand for the three considered discrete positions. In a further step, the parameter identification, respectively parameter updating, need to be automated for a continuous movement of the ram. Based on the structure of the equation system for the general machine model nonlinear optimization methods must be considered most likely.

However, independently from the finally used updating algorithm, the model has to be investigated with respect to real sensor signals. On the one hand, that relates to the accuracy of the adapted model compared to the real dynamic behavior of the machine structure. On the other hand, it has an effect on the parameter updating in terms of:

- Amount and kind of necessary sensor signals.
- Quality of measurement signals and sampling time.
- Real-time capability and data storage.

Furthermore, it has to be investigated how the machining process influences the online updating of the model parameters for two cases. First, how much disturbance the machining process causes in the updated model parameters of a general machine model, which should represent just low structural eigenfrequencies. And second, what possibilities do exist to model vibrations due to the machining process in the general machine model. In future, all these points will be investigated.

Acknowledgements

The author S. Apprich would like to thank the German Research Foundation (DFG) for financial support of the project within the Cluster of Excellence in Simulation Technology (EXC 310/2) at the University of Stuttgart.

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